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(71) Applicant: **HONDA GIKEN KOGYO KABUSHIKI KAISHA**
1-1, 2-chome
Minami-Aoyama
Minato-ku
Tokyo (JP)

(72) Inventor: **Sakai, Takafumi, c/o Kabushiki Kaisha Honda**
Gijutsu Kenkyusho,
No. 4-1, 1-chome, Chuo
Wako-shi,
Saitama-ken (JP)
Inventor: **Wakamatsu, Hideki, c/o Kabushiki Kaisha Honda**
Gijutsu Kenkyusho,
No. 4-1, 1-chome, Chuo
Wako-shi,
Saitama-ken (JP)
Inventor: **Haraguchi, Kenichi, c/o Kabushiki Kaisha Honda**
Gijutsu Kenkyusho,
No. 4-1, 1-chome, Chuo
Wako-shi,
Saitama-ken (JP)

(74) Representative: **Fincke, Karl Theodor, Dipl.-Phys. Dr. et al**
Patentanwälte
H. Weickmann, Dr. K. Fincke
F.A. Weickmann, B. Huber
Dr. H. Liska, Dr. J. Prechtel, Dr. B. Böhm,
Kopernikusstrasse 9
D-81679 München (DE)

(54) Control apparatus for hydraulically operated vehicular transmission.

(57) A control apparatus for a hydraulically operated vehicular transmission has a plurality of hydraulic engaging elements including a first hydraulic engaging element (C1) which is kept engaged in a plurality of transmission trains. The control apparatus has an accumulator (21) which is connected, via a branched oil passage (L14), to a working oil passage (L2) which is communicated with the first hydraulic engaging element (C1). It also has a control valve (22) which is disposed in the branched oil passage (L14) and which can be moved between a first position in which an upstream portion of the branched oil passage on a side of the first hydraulic engaging element (C1) and a downstream portion of the branched oil passage on a side of the accumulator (21) are brought into communication with each other and a second position in which the communication is shut off. It further has an oil discharge passage (LD) which is connected to a second hydraulic engaging element when the second hydraulic engaging element is disengaged. This oil discharge passage (LD) is arranged to be connected to the downstream portion of the branched oil passage (L14) in the second position of the control valve (22).

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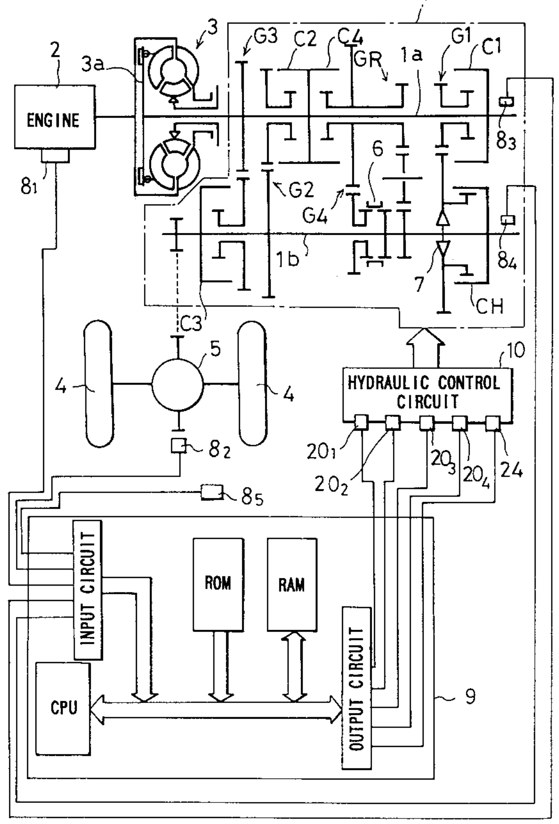
FIG.1


1
2
3
3a
ENGINE
8_1
G3
C2
C4
GR
1a
G1
C1
8_3
6
G2
G4
1b
7
8_4
CH
C3
5
4
4
10
HYDRAULIC CONTROL CIRCUIT
8_2
20_1
20_2
20_3
20_4
24
8_5
INPUT CIRCUIT
ROM
RAM
CPU
OUTPUT CIRCUIT
9

TECHNICAL FIELD

The present invention relates to a control apparatus for a hydraulically operated vehicular transmission which has a plurality of hydraulic engaging elements.

BACKGROUND ART

In a hydraulically operated transmission for use in a vehicle such as an automobile, it is normal practice to effect the speed changing or shifting by controlling the engaging or disengaging of a plurality of hydraulic engaging elements. In this case, unless an appropriate control is made of a simultaneous engagement of a hydraulic engaging element to be engaged and a hydraulic engaging element to be disengaged at the time of speed changing, there will occur speed-change shocks (i.e., shocks at the time of speed shifting). As a solution, the following arrangement is disclosed in Japanese Published Unexamined Patent Application No. 180756/1988. Namely, an accumulator is provided in parallel with each of the hydraulic engaging elements, and an oil discharge control valve is provided in an oil discharge passage that is connected to the hydraulic engaging element to be disengaged. The pressure decrease in the hydraulic engaging element to be disengaged is buffered or cushioned by the accumulator that is parallelly connected thereto and causes a simultaneous engagement to take place. Once the hydraulic oil pressure in the hydraulic engaging element to be engaged has increased to a certain degree, the oil discharge control valve is opened to rapidly decrease the hydraulic oil pressure in the hydraulic engaging element to be disengaged, thereby preventing an excessive simultaneous engagement.

If one accumulator must be provided for each of the hydraulic engaging elements as in the above-described conventional example, there is a disadvantage in that a large number of accumulators are required and consequently that the control apparatus becomes large in size.

By the way, in the above-described conventional example, there is interposed in the first-speed transmission train a one-way clutch which allows for overrunning of the output side. The first-speed transmission train is thus established by engaging the hydraulic engaging element for the first-speed transmission train and, while this hydraulic engaging element is kept engaged, other hydraulic engaging elements are respectively engaged to thereby establish the other transmission trains. As described above, in the transmission provided with a hydraulic engaging element that is kept engaged in a plurality of transmission trains, the accumulator for that particular hydraulic engaging element will always be kept in a condition of pressure accumulation.

Taking note of this point, the present invention has an object of providing a control apparatus in which the pressure decrease characteristics at the time of disengagement of other hydraulic engaging elements can be controlled, by the accumulator for the hydraulic engaging element that is always kept engaged, thereby making it possible to decrease the number of accumulators to be used.

DISCLOSURE OF THE INVENTION

In order to attain the above-described and other objects, the present invention is a control apparatus for a hydraulically operated vehicular transmission, the control apparatus having a plurality of hydraulic engaging elements including a first hydraulic engaging element which is kept engaged in a plurality of transmission trains. The control apparatus comprises an accumulator which is connected, via a branched oil passage, to a working oil passage which is communicated with the first hydraulic engaging element, a control valve which is disposed in the branched oil passage and which can be moved between a first position in which an upstream portion of the branched oil passage on a side of the first hydraulic engaging element and a downstream portion of the branched oil passage on a side of the accumulator are brought into communication with each other and a second position in which the communication is shut off, and an oil discharge passage which is connected to a second hydraulic engaging element when the second hydraulic engaging element is disengaged. The oil discharge passage is arranged to be connected to the downstream portion of the branched oil passage in the second position of the control valve.

At the time when the speed changing is effected by engaging a third hydraulic engaging element while disengaging the second hydraulic engaging element, the control valve is changed over from the first position to the second position. According to this operation, the accumulator that has already been accumulated in pressure in advance by the hydraulic oil supply to the first hydraulic engaging element, is communicated with the oil discharge passage via the control valve. The accumulated hydraulic oil is supplied from the accumulator to the oil discharge passage to thereby buffer or cushion a sudden pressure decrease in the second hydraulic engaging element. When the control valve is then returned from the second position to the first position at a predetermined time point (or timing) in a transient period of the speed changing, the communication between the accumulator and the oil discharge passage is shut off. The effect by the accumulator of buffering the pressure decrease is thus stopped and the hydraulic oil pressure in the second hydraulic engaging element is rapidly decreased.

The above-described predetermined time point for returning the control valve to the first position may be defined by using, as a parameter, an elapsed time from a time of a speed change judgement or a speed change order, a revolution speed of a member that is subjected to a change in revolution speed by the speed changing (for example, one or more of a revolution speed of an input shaft or an output shaft of the transmission and a revolution speed of an engine, and the vehicle speed), or a torque of the output shaft of the transmission. In this case, if the defined value of these parameters is set according to at least one of the engine load, the vehicle speed and the kind or mode of speed changing, the characteristics of pressure decrease, during the transient period of speed changing, in the second hydraulic engaging element to be disengaged can be more appropriately controlled depending on the operating conditions. The above-described predetermined time point may be defined by the hydraulic pressure in the third hydraulic engaging element to be engaged.

It is preferable to provide solenoid valve means to control the changeover of the control valve. The solenoid valve means preferably comprises a first solenoid valve which generates a hydraulic oil pressure to urge the control valve to the second position and a second solenoid valve which generates a hydraulic oil pressure to urge the control valve to the first position so that the control valve can be changed over between the first position and the second position via the solenoid valve means.

When a delay in operation of the control valve is taken into consideration, it is preferable to provide speed change expecting means to expect the time of the speed changing so that the control valve may be set to the second position prior to the outputting of the speed change judgement or the speed change order. As the speed change expecting means, a timer may be used which detects a time that is earlier by a predetermined time than the time of the speed change judgement or the speed change order. In this case, preferably the value of the timer is set according to at least one of the engine load, the vehicle speed and the kind or mode of speed changing.

BRIEF DESCRIPTION OF THE DRAWINGS

The above and other objects and the attendant advantages of the present invention will become readily apparent by reference to the following detailed description when considered in conjunction with the accompanying drawings wherein:

Fig. 1 is a system diagram showing a transmission in which the present invention is applied and the control system thereof;

Fig. 2 is a circuit diagram showing a hydraulic control circuit thereof;

Fig. 3 is an enlarged view of a manual valve and shift valves in the circuit of Fig. 2;

Fig. 4 is an enlarged view of a control valve; and

Figs. 5A and 5B are graphs showing change characteristics of clutch pressure at the time of speed changing, in which Fig. 5A is a graph showing the change characteristics at the time of upshifting from the second speed to the third speed and Fig. 5B is a graph showing the change characteristics at the time of downshifting from the third speed to the second speed.

DETAILED DESCRIPTION OF A PREFERRED EMBODIMENT

Referring to Fig. 1, numeral 1 denotes a transmission for effecting speed changing or shifting of four forward transmission trains and one reverse transmission train. Between an input shaft 1a which is connected to an engine 2 via a fluid torque converter 3 having a clutch 3a and an output shaft 1b which is connected via a differential gear 5 to driven wheels 4 of a vehicle, there are parallelly provided first-speed through fourth-speed forward transmission trains G1, G2, G3, G4 and one reverse transmission train GR. In the forward transmission trains there are interposed hydraulic engaging elements in the form of hydraulic clutches C1, C2, C3, C4, respectively, to selectively establish each of the forward transmission trains through engagement of each of the hydraulic clutches. The reverse transmission train GR is arranged to share the use of the fourth-speed hydraulic clutch C4 with the fourth-speed transmission train G4. The fourth-speed transmission train G4 and the reverse transmission train GR are thus selectively established by the changing over of a selector gear 6 which is provided on the output shaft 1b to the forward running position on the left-hand side in the drawing and to the reverse running position on the right-hand side therein.

In the first-speed transmission train G1 there is interposed a one-way clutch 7 which allows for overrunning of the output side. It is thus so arranged that, even in a condition in which the first-speed hydraulic clutch C1 is kept engaged, each of the transmission trains G2, G3, G4 of the second speed through the fourth speed can be established by engaging the second-speed through the fourth-speed hydraulic clutches C2, C3, C4. Further, there is provided a first-speed holding clutch CH which bypasses the one-way clutch 7. It is thus so arranged that the first-speed transmission train G1 can be established in a condition in which, by the engagement of the hydraulic clutch CH, the overrunning of the output side is not allowed, i.e., a condition in which the engine braking can be applied.

There are provided an electronic control circuit 9 and a hydraulic control circuit 10 for the above-described plurality of hydraulic clutches. The electronic control circuit 9 is made up of a microcomputer to which there are inputted signals from an engine sensor 8_1 for detecting a throttle opening, a revolution speed, a cooling water temperature or the like of the engine 2, a vehicle speed sensor 8_2 for detecting the speed of the vehicle based on the revolution speed of the differential gear 5, rotation sensors 8_3, 8_4 for detecting the revolution speeds of the input shaft 1a and the output shaft 1b, respectively, of the transmission 1, and a position sensor 8_5 for the manual valve which is to be described hereinafter. Speed changing is thus made by controlling a plurality of solenoid valves, which are described hereinafter, of the hydraulic control circuit 10 by the electronic control circuit 9.

The hydraulic control circuit 10 is provided, as shown in Figs. 2 and 3, with a hydraulic oil pressure source 11, a manual valve 12 which can be changed over to seven positions of "P" for parking, "R" for reverse running, "N" for neutral, "D" and "S" for automatic speed changing, "2" for holding the second speed and "L" for holding the first speed, three sets of No. 1 through No. 3 shift valves 13, 14, 15, and a servo valve 16 to which is connected a shift fork 6a of the selector gear 6 for changing over between the forward running and the reverse running.

In the "D" position of the manual valve 12, No. 1 oil passage L1 which is communicated with the hydraulic oil pressure source 11 is connected, via an annular groove 12a of the manual valve 12, to No. 2 oil passage L2 which is communicated with the first-speed hydraulic clutch C1. Pressurized hydraulic oil which is adjusted by a regulator valve 17 to a constant line pressure is always supplied from No. 1 oil passage L1 to the first-speed hydraulic clutch C1 via No. 2 oil passage L2. Further, the hydraulic oil is selectively supplied to the second-speed through the fourth-speed hydraulic clutches C2, C3, C4 via No. 3 oil passage L3 which is branched from No. 2 oil passage L2 and via the above-described shift valves 13, 14, 15.

Here, No. 1 shift valve 13 is urged to the left-hand side by a spring 13a and No. 2 shift valve 14 and No. 3 shift valve 15 are urged to the right-hand side by a spring 14a, 15a, respectively. A modulator pressure (a constant pressure which is lower than the line pressure) from a modulator valve 18 which is connected to No. 1 oil passage L1 is inputted, via No. 5 oil passage L5 that is communicated via an orifice 19_1 with No. 4 oil passage L4 on the output side of the modulator valve 18, to a left end oil chamber 13b of No. 1 shift valve 13 and to a right-hand side oil chamber 15b of No. 3 shift valve 15, respectively. The modulator pressure is also inputted, via No. 6 oil passage L6 which is communicated via an orifice 19_2 with No. 4 oil passage L4, to a right end oil chamber 13c of No. 1 shift valve 13 and to a right end oil chamber 14b of No. 4 shift valve 14, respectively. There are further provided in No. 5 oil passage L5 a normally-closed type No. 1 solenoid valve 20_1 which opens No. 5 oil passage L5 to atmosphere and in No. 6 oil passage L6 a normally-closed type No. 2 solenoid valve 20_2 which opens No. 6 oil passage L6 to atmosphere. By the opening and closing of these two solenoid valves 20_1, 20_2, these shift valves 13, 14, 15 are changed over in the following manner in accordance with each of the transmission trains.

Namely, at the first speed, No. 1 solenoid valve 20_1 is closed and No. 2 solenoid valve 20_2 is opened. According to this operation, the modulator pressure is inputted to the left end oil chamber 13b of No. 1 shift valve 13 and to the right-hand side oil chamber 15b of No. 3 shift valve 15 respectively, and the right end oil chamber 13c of No. 1 shift valve 13 and the right end oil chamber 14b of No. 2 shift valve 14 are opened to atmosphere. No. 1 shift valve 13 and No. 2 shift valve 14 are thus changed over to right-hand side position and No. 3 shift valve 15 is changed over to the left-hand side position. In this condition, No. 7 oil passage L7 which is communicated with No. 2 hydraulic clutch C2 is connected, via an annular groove 14c of No. 2 shift valve 14, to No. 8 oil passage L8 which leads to No. 1 shift valve 13, and No. 8 oil passage L8 is connected, via an annular groove 13d of No. 1 shift valve 13, to No. 9 oil passage L9 that is connected to an open-to-atmosphere groove 12b of the manual valve 12 in the "D" position thereof, thereby discharging the hydraulic oil from the second-speed hydraulic clutch C2. No. 10 oil passage L10 which is communicated with the third-speed hydraulic clutch C3 is connected, via an annular groove 15c of No. 3 shift valve 15, to an oil discharge passage LD, thereby discharging the hydraulic oil from the third-speed hydraulic clutch C3. No. 11 oil passage L11 which is communicated with the fourth-speed hydraulic clutch C4 is connected, via an annular groove 15d of No. 3 shift valve 15, to No. 12 oil passage L12 which leads to No. 2 shift valve 14, and No. 12 oil passage L12 is connected, via an annular groove 14d of No. 2 shift valve 14, to No. 13 oil passage L13 which leads to No. 3 shift valve 15. No. 13 oil passage L13 is connected, via an annular groove 15e of No. 3 shift valve 15, to an oil discharge port 15g thereof, thereby discharging the hydraulic oil from the fourth-speed hydraulic clutch C4. Therefore, it is only the first-speed hydraulic clutch C1 that is supplied with the hydraulic oil, thereby establishing the first-speed transmission train G1.

At the second speed, both No. 1 solenoid valve 20_1 and No. 2 solenoid valve 20_2 are opened. According to this operation, No. 1 shift valve 13 is changed over to the left-hand side position and No. 2 shift valve 14 and No. 3 shift valve 15 are changed over to the right-hand side position. In this condition, No. 3 oil passage L3 is connected to No. 8 oil passage L8 via the annular groove 13d of No. 1 shift valve 13, and No. 8 oil passage L8 is connected to No. 7 oil passage L7 via the annular groove 14c of No. 2 shift valve 14, thereby supplying the hydraulic oil to the second-speed hydraulic clutch C2. On the other hand, No. 10 oil passage L10 which is communicated with the third-speed hydraulic clutch C3 is connected to the oil discharge passage LD via the route of the annular groove 15c of No. 3 shift valve 15, No. 12 oil passage L12, the annular groove 14d of No. 2 shift valve 14, No. 13 oil passage L13 and the annular groove 15e of No. 3 shift valve 15, thereby discharging the hydraulic oil from the third-speed hydraulic clutch C3. Further, No. 11 oil passage L11 which is communicated with the fourth-speed hydraulic clutch C4 is connected, via the annular groove 15d of No. 3 shift valve 15, to an oil discharge port 15h of No. 3 shift valve 15, thereby discharging the hydraulic oil from the fourth-speed hydraulic clutch C4. The second-speed transmission train G2 is therefore established.

At the third speed, No. 1 solenoid valve 20_1 is opened and No. 2 solenoid valve 20_2 is closed. According to this operation, No. 1 shift valve 13 is kept in the left-hand side position, No. 3 shift valve 15 is kept in the right-hand side position and No. 2 shift valve 14 is changed over to the left-hand side position by the input of the modulator pressure to the right end oil chamber 14b of No. 2 shift valve 14. In this condition, like at the second speed, No. 8 oil passage L8 which is connected to No. 3 oil passage L3 via the annular groove 13d of No. 1 shift valve 13 is connected to No. 12 oil passage L12 via the annular groove 14d of No. 2 shift valve 14. Here, like at the second speed, since No. 12 oil passage L12 is connected to No. 10 oil passage L10 via the annular groove 15c of No. 3 shift valve 15, the hydraulic oil is supplied to the third-speed hydraulic clutch C3. On the other hand, No. 7 oil passage L7 which is communicated with the second-speed hydraulic clutch C2 is connected to the oil discharge passage LD via the route of the annular groove 14c of No. 2 shift valve 14, No. 13 oil passage L13 and the annular groove 15e of No. 3 shift valve 15, thereby discharging the hydraulic oil from the second-speed hydraulic clutch C2. Further, No. 11 oil passage L11 which is communicated with the fourth-speed hydraulic clutch C4 is connected, like at the second speed, to the oil discharge port L15h of No. 3 shift valve 15 via the annular groove 15d of No. 3 shift valve 15, thereby discharging the hydraulic oil from the fourth-speed hydraulic clutch C4. The third-speed transmission train G3 is therefore established.

At the fourth speed, No. 1 solenoid valve 20_1 and No. 2 solenoid valve 20_2 are both closed. According to this operation, the modulator pressure is inputted to the left end oil chamber 13b of No. 1 shift valve 13, but this rightward urging force is balanced with the modulator pressure to be inputted to the right end oil chamber 13c of No. 1 shift valve 13, with the result that No. 1 shift valve 13 is held in the left-hand side position of No. 1 shift valve 13 due to the force of the spring 13a. No. 2 shift valve 14 is also held in the left-hand side position like at the third speed but, on the other hand, No. 3 shift valve 15 is changed over to the left-hand side position by the input of the modulator pressure to the right-hand side oil chamber 15b of No. 3 shift valve 15. In this condition, No. 12 oil passage L12 which is communicated with No. 3 oil passage L3 via No. 8 oil passage L8 is connected to No. 11 oil passage L11 via the annular groove 15d of No. 3 shift valve 15, thereby supplying the hydraulic oil to the fourth-speed hydraulic clutch C4. On the other hand, No. 10 oil passage L10 which is communicated with the third-speed hydraulic clutch C3 is connected to the oil discharge passage LD via the annular groove 15c of No. 3 shift valve 15, thereby discharging the hydraulic oil from the third-speed hydraulic clutch C3. Furthermore, No. 13 oil passage L13 which is connected, via the annular oil groove 14c of No. 2 shift valve 14, to No. 7 oil passage L7 which is communicated with the second-speed hydraulic clutch C2, like at the third speed, is connected to the oil discharge port 15g of No. 3 shift valve 15, thereby discharging the hydraulic oil from the second-speed hydraulic clutch C2. The fourth-speed transmission train G4 is therefore established.

The opening and closing of No. 1 solenoid valve 20_1 and No. 2 solenoid valve 20_2 and the changeover positions of No. 1 through No. 3 shift valves 13, 14, 15 can be summarized as shown in Table 1. In the "D" position of the manual valve 12, No. 1 solenoid valve 20_1 and No. 2 solenoid valve 20_2 are controlled for opening or closing by the electronic control circuit 9 according to the speed-change characteristics to be set based on the speed of the vehicle and throttle opening, thereby effecting the automatic speed changing of the first speed through the fourth speed.

Table 1

	No. 1 solenoid valve	No. 2 solenoid valve	No. 1 shift valve	No. 2 shift valve	No. 3 shift valve
First speed	closed	open	right	right	left
Second speed	open	open	left	right	right
Third speed	open	closed	left	left	right
Fourth speed	closed	closed	left	left	left

By the way, to No. 2 oil passage L2 which is communicated with the first-speed hydraulic clutch C1, there is connected an accumulator 21 via No. 14 oil passage L14 which is branched from No. 2 oil passage L2. In this No. 14 oil passage L14 there is disposed a control valve 22, as clearly shown in Fig. 4. This control valve 22 can be changed over between the left-hand side position in which the upstream portion and the downstream portion of No. 14 oil passage L14 are brought into communication with each other and the right-hand side position in which the above-described communication is cut off. There is provided in this control valve 22 a port that is communicated with the above-described oil discharge passage LD. In the right-hand side position of the control valve 22, the downstream portion of No. 14 oil passage L14 that is communicated with the accumulator 21 is arranged to be connected to the oil discharge passage LD. In the left-hand side position of the control valve 22, the oil discharge passage LD is arranged to be connected to an oil discharge port 22a of the control valve 22.

The control valve 22 is urged to the left by a spring 22b and is further provided with a left end oil chamber 22c and a right end oil chamber 22d. The left end oil chamber 22c is arranged to receive an input of the modulator pressure from the modulator valve 18 via No. 15 oil passage L15 which is connected to No. 4 oil passage L4 via an orifice 19_3. The right end oil chamber 22d is arranged to receive an input of the modulator pressure via No. 16 oil passage L16 which is communicated with No. 4 oil passage L4 via an orifice 19_4. In No. 15 oil passage L15 there is disposed a normally-closed type No. 3 solenoid valve 20_3 which opens No. 15 oil passage L15 to atmosphere. In No. 16 oil passage L16 there is disposed a normally-closed type No. 4 solenoid valve 20_4 which opens No. 16 oil passage L16 to atmosphere. When No. 3 solenoid valve 20_3 is closed, the control valve 22 is changed over to the right-hand side position by the input of the modulator pressure to the left end oil chamber 22c. When No. 4 solenoid valve 20_4 is closed even when No. 3 solenoid valve 20_3 is kept closed, the modulator pressure is inputted to the right end oil chamber 22d, with the result that the urging forces by the modulator pressure on the right and the left oil chambers 22c, 22d are well balanced and, consequently, the control valve 22 is changed over to the left-hand side position by the force of the spring 22b.

Here, the discharging of the hydraulic oil at the time of the speed changing via the oil discharge passage LD from a hydraulic clutch to be disengaged takes place at the time of upshifting from the second speed to the third speed, upshifting from the third speed to the fourth speed, downshifting from the third speed to the second speed, and downshifting from the third speed to the first speed. At the time of these speed changing, except at the time of downshifting from the third speed to the first speed when the one-way clutch 7 works, the control valve 22 is changed over to the right-hand side position up to a predetermined time point (or timing) of the transient period of the speed changing to connect the accumulator 21 to the oil discharge passage LD. The accumulated hydraulic oil in the accumulator 21 is thus supplied to the oil discharge passage LD to buffer the decrease or drop in the pressure of the hydraulic clutch to be disengaged. Thereafter, the control valve 22 is changed over to the left-hand side position to connect the oil discharge passage LD to the oil discharge port 22a. The hydraulic oil from the hydraulic clutch to be disengaged is thus made to be discharged also from the oil discharge port 22a, in addition to an oil discharge port LDa with an orifice of the oil discharge passage LD. The pressure decrease or drop characteristics of the hydraulic clutch to be disengaged is thus made to be accompanied with slowness or rapidity so that an appropriate simultaneous engagement thereof with the hydraulic clutch to be engaged can be obtained.

By the way, the hydraulic oil pressure to a hydraulic clutch to be engaged is adjustable by a pressure adjusting valve 23 which is interposed in No. 3 oil passage L3. There is inputted to this pressure adjusting valve 23, via No. 17 oil passage L17, an output hydraulic oil pressure from a hydraulic oil pressure control valve 24 which is made up of a linear solenoid valve to be controlled by the electronic control circuit 9. The pressure increase characteristics in the hydraulic clutch to be engaged are controlled by the hydraulic oil pressure control valve 24 via the pressure adjusting valve 23.

In the "D" position of the manual valve 12, No. 18 oil passage L18 is connected to No. 1 oil passage L1 via the annular groove 12a of the manual valve 12. No. 19 oil passage L19 which is connected via a servo valve 25, which is to be described hereinafter, to No. 18 oil passage L18 is connected to No. 20 oil passage L20 which is communicated with a back pressure chamber of the accumulator 21 via an annular groove 14e of No. 2 shift valve 14 that is in the left-hand side position at the time of the third speed and the fourth speed. The internal pressure in the accumulator 21 thus becomes equal to the line pressure. On the other hand, at the second speed, by the changeover of No. 2 shift valve 14 to the right-hand side position, No. 20 oil passage L20 is connected, via the annular groove 14e of No. 2 shift valve 14, to No. 21 oil passage L21 which leads to No. 1 shift valve 13. No. 21 oil passage L21 is connected to an oil discharge port 13f of No. 1 shift valve 13 via an annular groove 13e of No. 1 shift valve 13 that is in the left-hand side position. The internal pressure of the back pressure chamber of the accumulator 21 therefore becomes equal to the atmospheric pressure.

In this manner, the pressure decrease in the second-speed hydraulic clutch C2 to be disengaged which is connected to the oil discharge passage LD at the time of upshifting from the second speed to the third speed, and the pressure decrease in the third-speed hydraulic clutch C3 to be disengaged which is connected to the oil discharge passage LD at the time of upshifting from the third speed to the fourth speed are buffered in a relatively high hydraulic oil pressure range as shown in Fig. 5A because the back pressure in the accumulator 21 is high. By the pressure increase in the hydraulic clutch to be engaged, there occurs a condition in which the hydraulic clutch to be engaged and the hydraulic clutch to be disengaged are simultaneously engaged. The degree of this simultaneous engagement can be judged by the change in the revolution speed of such a member as the input shaft, the output shaft, the engine or the like whose revolution speed varies by the speed changing,. For example, if the slip ratio S (= r x Nout ÷ Nin) of the hydraulic clutch to be disengaged is obtained by the gear ratio r of the transmission train, the revolution speed Nin of the input shaft 1a and the revolution speed Nout of the output shaft 1b of the transmission, the slip ratio S increases with the decrease in the value Nin through the simultaneous engagement. In addition, since the output shaft torque T of the transmission also decreases through the simultaneous engagement, it can be used as a parameter in judging the degree of simultaneous engagement. Therefore, if the control valve 22 is changed over to the left-hand side position when either of the slip ratio S or the output shaft torque T has changed to a predetermined value Ss, Ts and brings the oil discharge passage LD into communication with the oil discharge port 22a to rapidly decrease or drop the hydraulic oil pressure in the hydraulic clutch to be disengaged, the simultaneous engagement can be appropriately controlled and a smooth speed changing without the occurrence of shocks can be effected. By the way, it is preferable to variably set the above-described predetermined values Ss, Ts depending on the engine load, vehicle speed and the kind or mode of the speed changing. Further, the gear ratio r to calculate the slip ratio S is changed, at the time of change in the slip ratio S, from the gear ratio of the transmission train to be disengaged to the gear ratio of the transmission train to be engaged, to make it an index of judging the degree of engagement of the hydraulic clutch to be engaged. An arrangement may also be made such that the control valve 22 is changed over to the left-hand side position when the hydraulic oil pressure in the hydraulic clutch to be engaged has increased to a predetermined pressure.

At the time of downshifting from the third speed to the second speed, since the back pressure in the accumulator 21 becomes low, the pressure decrease in the third-speed hydraulic clutch C3 to be disengaged is buffered in a relatively low hydraulic oil pressure range as shown in Fig. 5B. Therefore, the engine races to an appropriate degree and a smooth speed changing is effected. At a lapse of a predetermined time t1 from the outputting of an order for speed changing, the control valve 22 is changed over to the left-hand side position. The control valve 22 may also be changed over to the left-hand side position based on the slip ratio S as described above.

At the time of starting the speed changing it is necessary for the control valve 22 to have been changed over to the right-hand side position. Now, when the response time lag from the time of order for changeover of the control valve 22 to the time when the control valve 22 is actually changed over is taken into consideration, it is preferable to output the order for changeover of the control valve 22 at a predetermined time ahead of the time of outputting the order for the speed changing. Specifically, the timing of the next speed changing is expected based on the changes in the throttle opening and in the vehicle speed, and the changeover order is outputted at a time that is earlier by a predetermined time t2 than the timing of the next speed changing, thereby changing over the control valve 22 to the right-hand side position. It is preferable to variably set this time t2 depending on the divisional changes in the throttle opening and the vehicle speed, kind or mode of expected speed changing, or the like.

The order to change over the control valve 22 to the right-hand side position is made by outputting order signals to close No. 3 solenoid valve 20_3 and to open No. 4 solenoid valve 20_4. Here, No. 3 solenoid valve 20_3 also serves the function of a solenoid valve to change over a shift valve 27 which is provided in a conventional clutch control circuit 26, the shift valve 27 being for controlling the clutch 3a for the fluid torque converter 3 by using, as the working fluid, the hydraulic oil to be supplied from the regulator valve 17 via No. 22 oil passage L22. When No. 3 solenoid valve 20_3 is closed, the shift valve 27 is changed over to the leftward position in which the clutch 3a is disengaged. Therefore, at the time of the above-described speed changing in which the control valve 22 is changed over to the right-hand side position, the clutch 3a is disengaged and, due to the slippage in the fluid torque converter 3, the occurrence of shocks at the time of speed changing is more effectively restrained or kept under control. In this case, it is better to keep the clutch 3a disengaged until the speed changing has been finished. Therefore, in the present embodiment, it is so arranged, as described above, that the control valve 22 can be changed over to the left-hand side position by closing No. 4 solenoid valve 20_4 even when No. 3 solenoid valve 20_3 is kept closed. No. 4 solenoid valve 20_4 also serves the function of a solenoid valve which controls a control valve 28 and a timing valve 29 which are provided in the clutch control circuit 26 to adjust the engaging force of the clutch 3a.

In the position of the shift valve 27 in which the clutch 3a is disengaged, a branch oil passage L19a of No. 19 oil passage L19 is connected to No. 23 oil passage L23 which leads to the control valve 22. By the changing over of the control valve 22 to the left-hand side position, the line pressure is inputted from No. 23 oil passage L23 to the pressure adjusting valve 23 via No. 24 oil passage L24. The output hydraulic oil pressure of the pressure adjusting valve 23, i.e., the hydraulic oil pressure of the hydraulic clutch to be engaged is thus boosted depending on the decrease in the hydraulic oil pressure in the hydraulic clutch to be disengaged.

In order to decrease or alleviate the in-gear shocks, i.e., shocks at the time of gear engagement, when the gear is engaged to start the vehicle by changing over the manual valve 12 from the "P" or "N" position to the "D" position, a squat control is made in which the second-speed transmission train G2 is established first by changing over No. 1 shift valve 13 to the left-hand side position and No. 2 shift valve 14 to the right-hand side position and then the first-speed transmission train G1 is established by changing over No. 1 shift valve 13 to the right-hand side position. At this time, it is necessary to prevent the hydraulic oil pressure in the first-speed hydraulic clutch C1 from rising earlier than the hydraulic oil pressure in the second-speed hydraulic clutch C2. For that purpose, No. 2 oil passage L2 is provided with a small-diameter orifice 30 which is located on a downstream side of the branched portion of No. 3 oil passage L3. Furthermore, No. 2 oil passage L2 is provided with a pair of bypass passages L2a, L2b which bypass the orifice 30. There is interposed in one of them, i.e., in No. 1 bypass passage L2a, a check valve 31 which allows for the oil supply from the upstream side to the downstream side and, in the other of them, i.e., in No. 2 oil passage L2b, a check valve 32 which allows for the oil supply from the downstream side to the upstream side. Further, by the changing over of No. 1 shift valve 13 both bypass passages L2a, L2b are arranged to be selectively communicated via an annular groove 13g of No. 1 shift valve 13. Until No. 1 shift valve 13 is changed over, by the squat control, from the left-hand side position to the right-hand side position to thereby shift down from the second speed to the first speed, the hydraulic oil is slowly supplied to No. 1 hydraulic clutch C1 only via the orifice 30. When the downshifting has been effected, No. 1 bypass passage L2a is opened to communication and, by the hydraulic oil supply via the bypass passage L2a, No. 1 hydraulic clutch C1 is arranged to quickly rise in the hydraulic oil pressure.

Since the back pressure in the accumulator 21 at the second speed is in a low pressure as described above, the boosting of the hydraulic pressure in the first-speed hydraulic clutch C1 is buffered in a relatively low hydraulic oil pressure range. Therefore, the setting up or rise in the hydraulic oil pressure in the first-speed hydraulic clutch C1 at the time of establishing the second speed by the squat control is more effectively restrained. At the first speed, No. 21 oil passage L21 which is connected, via the annular groove 14e of No. 2 shift valve 14 that is in the right-hand side position, to No. 20 oil passage L20 which is communicated with the back pressure chamber of the accumulator 21, is connected to No. 3 oil passage L3 via the annular groove 13e of No. 1 shift valve 13 that is in the right-hand side position. Therefore, there will occur a condition in which the back pressure in the accumulator 21 can be controlled by the hydraulic oil pressure control valve 24 via the pressure adjusting valve 23. The pressure increase characteristics of No. 1 hydraulic clutch C1 after the downshifting from the second speed to the first speed can appropriately be controlled without giving rise to in-gear shocks.

The above-described No. 2 bypass passage L2b functions to control the pressure decrease characteristics of the first-speed hydraulic clutch C1 at the time of changing over from the "D" position to the "N" position. Its details are explained hereinafter.

Explanations have so far been made about the arrangement of the hydraulic oil circuit in the "D" position of the manual valve 12. Similar arrangement in the hydraulic oil circuit as in the "D" position also applies to the "S" position, in which automatic speed changing between the first speed through the fourth speed is effected according to speed change characteristics that are different from those in the "D" position.

In the "2" position of the manual valve 12, No. 1 oil passage L1 is connected to No. 2 oil passage L2 via the annular groove 12a of the manual valve 12 and is also connected, via the annular grooves 12a, a connection passage 12c and an annular groove 12d, to No. 9 oil passage L9. The line pressure is thus inputted to No. 1 shift valve 13 via No. 9 oil passage L9 and also the output hydraulic oil pressure of the pressure adjusting valve 23 is inputted to No. 1 shift valve 13 via No. 3 oil passage L3. To No. 8 oil passage L8 which leads to No. 2 shift valve 14 there are connected No. 9 oil passage L9 in the right-hand side position of No. 1 shift valve 13 via the annular groove 13d of No. 1 shift valve 13, as well as No. 3 oil passage L3 in the left-hand side position thereof. When No. 8 oil passage L8 is connected to No. 7 oil passage L7 via the annular groove 14c of No. 2 shift valve 14 by changing over No. 2 shift valve 14 to the right-hand side position, the hydraulic oil is supplied to the second-speed hydraulic clutch C2 in whichever, i.e., right or left, position No. 1 shift valve 13 may be positioned, thereby establishing the second-speed transmission train G2. By the way, if the manual shift valve 12 is changed over to the "2" position to thereby suddenly downshift to the second speed while running at a high speed, there sometimes occurs overrunning of the engine or large speed change shocks. Therefore, the following procedure is followed. Namely, when speed changing has been made to the "2" position at above a predetermined vehicle speed, No. 2 shift valve 14 is changed over to the left-hand side position to connect No. 8 oil passage L8 to No. 12 oil passage L12 via the annular groove 14d of No. 2 shift valve 14. The hydraulic oil is thus supplied to the third-speed hydraulic clutch C3 via No. 3 shift valve 15 to thereby establish the third-speed transmission train G3. When the vehicle speed has once been reduced below a predetermined speed, No. 2 shift valve 14 is changed over to the right-hand side position to thereby downshift to the second speed. No. 3 shift valve 15 is provided with a left end oil chamber 15i to which is connected No. 25 oil passage L25 which extends from the servo control valve 25. No. 9 oil passage L9 is connected to No. 25 oil passage L25 via a shuttle valve 33. In the "2" position of the manual valve 12 the line pressure is inputted to the oil chamber 15i from No. 9 oil passage L9 via No. 25 oil passage L25. No. 3 shift valve 15 is restrained to the right-hand third-speed position in which No. 10 oil passage L10 which is communicated with the third-speed hydraulic clutch C3 is connected to No. 12 oil passage L12 via the annular groove 15c of No. 3 shift valve 15. Even if No. 1 solenoid valve 20_1 is closed, No. 3 shift valve 15 will not be changed over to the left-hand fourth-speed position.

In the "L" position of the manual valve 12, No. 1 oil passage L1 is connected to No. 2 oil passage 12 via the annular groove 12a of the manual valve 12 and also connected to No. 26 oil passage L26 which is communicated with a left end oil chamber 14f of No. 2 shift valve 14 via the annular groove 12a, the connecting passage 12c and the annular groove 12d. No. 2 shift valve 14 is thus restrained to the right-hand second-speed position. Further, No. 27 oil passage L27 which extends from No. 1 shift valve 13 is connected to No. 28 oil passage L28 which is communicated, via an annular groove 12e of the manual valve 12, with the first-speed holding clutch CH. When No. 1 shift valve 13 is changed over to the right-hand side position, No. 3 oil passage L3 is connected to No. 27 oil passage L27 via the annular groove 13e of No. 1 shift valve 13. The hydraulic oil is thus supplied to the first-speed holding clutch CH, thereby establishing the first-speed transmission train G1 in a condition in which the engine braking can be applied. When a changeover is made to the "L" position at above a predetermined vehicle speed, No. 1 shift valve 13 is changed over to the left-hand side position to connect No. 3 oil passage L3 to No. 8 oil passage L8. The hydraulic oil is thus supplied to the second-speed hydraulic clutch C2 via No. 2 shift valve 14 that is restrained to the right second-speed position, thereby establishing the second-speed transmission train G2. When the vehicle speed has been decreased below a predetermined speed, No. 1 shift valve 13 is changed over to the right-hand side position to thereby downshift to the first speed.

In the "R" position of the manual valve 12, No. 1 oil passage L1 is connected, via the annular groove 12a of the manual valve 12, to No. 29 oil passage L29 which leads to No. 1 sift valve 13. Further, a branch oil passage L11a which is branched, via a shuttle valve 34, from No. 11 oil passage L11 which is communicated with the fourth-speed hydraulic clutch C4 is connected, via the annular groove 12e of the manual valve 12, to No. 30 oil passage L30 which extends from No. 2 shift valve 14. In the "R" position, No. 1 shift valve is changed over to the left-hand side position. According to this operation, No. 29 oil passage L29 is connected, via an annular groove 13h of No. 1 shift valve 13, to No. 31 oil passage L31 which is communicated with a left end oil chamber 16a of the servo valve 16. The servo valve 16 is thus moved to the right by the line pressure that is inputted via No. 31 oil passage L31, with the result that the selector gear 6 is changed over to the right-hand reverse-running position. In the reverse-running position, No. 31 oil passage L31 is connected, via that annular groove 16b of the servo valve 16 which opens into the oil chamber 16a, to No. 32 oil passage L32 which leads to No. 2 shift valve 14. No. 32 oil passage L32 is connected to No. 30 oil passage L30 via No. 2 shift valve 14, and the hydraulic oil is supplied to the fourth-speed hydraulic clutch C4, via the branch oil passage 11a, thereby establishing the reverse running transmission train GR. By the way, No. 32 oil passage L32 is branched, at its downstream side, into two branch oil passages L32a, L32b. The hydraulic oil pressure in the branch oil passage L32b is made adjustable by the pressure adjusting valve 23. No. 30 oil passage L30 is selectively connected, via an annular groove 14g of No. 2 shift valve 14, to one L32a of the branched oil passages at the left-hand side position thereof 14, and to the other L32b thereof at the right-hand side position thereof 14. It is thus so arranged that the line pressure and the output pressure of the pressure adjusting valve 23 can be selected as the working pressure of the fourth-speed hydraulic clutch C4.

When the manual valve 9 is changed over from the "R" position to the forward-running positions of "D", "S", "2" and "L", the line pressure is inputted to the right-hand oil chamber 16c of the servo valve 16 via No. 18 oil passage L18, the servo control valve 25 and No. 19 oil passage 19. The servo valve 16 is thus moved to the left to change over the selector gear 6 to the left forward-running position. In this case, if the changeover is made from the "R" position to the forward-running position in a condition in which the driving wheels are slipping, the selector gear 6 sometimes moves to the left while the output shaft 1b is running through inertia in the opposite direction, with the result that a smooth changeover of the selector gear 6 to the forward-running position cannot sometimes be made. As a solution, the following arrangement has been made. Namely, the hydraulic oil pressure of the first-speed hydraulic clutch C1 is caused to be applied to the right end oil chamber of the servo control valve 25 via a branched oil passage L2c of No. 2 oil passage L2. When the hydraulic oil pressure in the first-speed hydraulic clutch C1 has increased to a certain degree and, due to the engaging force of the first-speed hydraulic clutch C1, the reverse running of the output shaft 1b has been braked, the servo control valve 25 is moved to the left against a spring 25a. No. 18 oil passage L18 is thus connected to No. 19 oil passage L19, thereby moving the servo valve 16 to the left. The hydraulic oil pressure of No. 31 oil passage L31 is also caused to be applied to the left end of the servo control valve 25 so that, at the time of reverse running, the servo control valve 25 can surely be returned to the right-hand side position. Furthermore, there is provided No. 33 oil passage L33 which is communicated, at the reverse-running position of the servo valve 16, with the right-hand side oil chamber 16c of the servo valve 16 via a notched groove 16d thereof. No. 25 oil passage L25 which is communicated with the left end oil chamber 15i of No. 3 shift valve 15 is arranged to be changed over, in the right-hand side position of the servo control valve 25, to No. 18 oil passage L18 and, in the left-hand side position of the servo control valve 25, to No. 33 oil passage L33. According to this arrangement, in case the selector gear 6 is not changed over to the forward-running position when a changeover has been made from the "R" position to the "D" or "S" position because the servo control valve 25 is restrained to the right-hand side position, or else, even when the servo control valve 25 has been changed over to the left-hand side position, because the servo valve 16 is restrained to the reverse-running position, the line pressure is inputted to the left end oil chamber 15i of No. 3 shift valve 15 from No. 18 oil passage L18 or No. 33 oil passage L33 via No. 25 oil passage L25. Therefore, No. 3 shift valve 15 is restrained to the right-hand third-speed position, and it becomes impossible to supply the hydraulic oil to the forth-speed hydraulic clutch C4. The establishment of the reverse running transmission train GR at the "D" or "S" position is thus prevented.

In the "N" position of the manual valve 12, the communication between No. 1 oil passage L1 and the other oil passages is cut off. By the way, when the vehicle is stopped in a condition in which the manual valve 12 is in the "D" position, the driving unit made up of the engine and the transmission displaces in posture while elastically deforming engine mounting elements by the driving reaction forces due to the power transmission via the first-speed transmission train G1. When the manual valve 12 is changed over at this time to the "N" position to attain the neutral condition, the displaced driving unit returns to the normal position, giving rise to a problem of shocks (off-gear shocks or shocks at the time of gear disengagement). In the "N" position, No. 2 oil passage L2 is opened to atmosphere via the annular groove 12d of the manual valve 12 and the connection passage 12c, and the hydraulic oil is discharged from the first-speed hydraulic clutch C1. At this time, if the hydraulic oil pressure in the first-speed hydraulic clutch C1 is gradually decreased by restricting the oil discharge by means of the orifice 30 in No. 2 oil passage L2, the above-described off-gear shocks by a rapid releasing of the driving force can be prevented. However, in case the viscosity of the hydraulic oil is high at a low temperature, the releasing of the first-speed hydraulic clutch C1 will become too slow if the oil discharge is restricted by the orifice 30. As a result, it will cause the dragging of the hydraulic clutch C1 and badly affects the durability of the hydraulic clutch C1. In this case, if No. 1 shift valve 13 is changed over to the left-hand side position, the above-described second bypass passage L2b is opened to communication to discharge the hydraulic oil by bypassing the orifice 30. Therefore, there will occur no such disadvantage as an excessive delay in the disengagement of the first-speed hydraulic clutch C1.

In the "N" position, the electric power supply to the splenoid valves is normally cut off to save the electric power consumption. In such a case, No. 1 shift valve 13 may be changed over to the left-hand side position by the closing operation of the normally-closed type No. 1 and No. 2 solenoid valves 20_1, 20_2. When the off-gear shocks become the problem, however, No. 1 shift valve 13 is held to the right-hand side position for a predetermined period of time from the time of changeover from the "D" position to the "N"position to shut off the second bypass passage L2b of No. 2 oil passage L2, thereby restricting the oil discharge through the orifice 30.

In the "P" position of the manual valve 12, No. 1 oil passage L1 is connected, like in the "R" position, to No. 29 oil passage L29 and changes over the servo valve 16 to the reverse-running position by the inputting of the line pressure thereto via No. 31 oil passage L31. However, since No. 30 oil passage L30 is not connected to the bypass passage L11a in the "P" position, the reverse transmission train GR is not established.

As can be seen from the above description, according to the present invention, by means of the accumulator for the first hydraulic engaging element that is always kept engaged, the pressure decrease characteristics of the remaining hydraulic engaging elements at the time of speed changing can be controlled. It is therefore not necessary to provide an accumulator to be exclusively used for each of the hydraulic engaging elements. The number of accumulators to be used can thus be decreased and the apparatus can be made small in size and light in weight.

Claims

1. A control apparatus for a hydraulically operated vehicular transmission, said control apparatus having a plurality of hydraulic engaging elements including a first hydraulic engaging element (C1) which is kept engaged in a plurality of transmission trains, said control apparatus comprising:

an accumulator (21) which is connected, via a branched oil passage (L14), to a working oil passage (L2) which is communicated with said first hydraulic engaging element (C1);

a control valve (22) which is disposed in said branched oil passage (L14) and which can be moved between a first position in which an upstream portion of the branched oil passage on a side of said first hydraulic engaging element (C1) and a downstream portion of the branched oil passage on a side of said accumulator (21) are brought into communication with each other and a second position in which said communication is shut off; and

an oil discharge passage (LD) which is connected to a second hydraulic engaging element when said second hydraulic engaging element is disengaged, said oil discharge passage (LD) being arranged to be connected to said downstream portion of said branched oil passage (L14) in said second position of said control valve (22).

2. A control apparatus according to claim 1, wherein said control valve (22) is set to said second position up to a predetermined time point of a transient period of speed changing and is thereafter set to said first position.

3. A control apparatus according to claim 1 or 2, wherein a movement of said control valve (22) is controlled by solenoid valve means.

4. A control apparatus according to claim 3, wherein said solenoid valve means comprises a first solenoid valve (20_3) which generates a hydraulic oil pressure to urge said control valve (22) from said first position to said second position, and a second solenoid valve (20_4) which generates a hydraulic oil pressure to urge said control valve (22) from said second position to said first position.

5. A control apparatus according to any one of claims 2 through 4, wherein said predetermined time point is defined by an elapsed time from a time of a speed change judgement or a speed change order.

6. A control apparatus according to claim 5, wherein a defined value of said elapsed time is set according to at least one of an engine load, a vehicle speed and a kind or mode of speed changing.

7. A control apparatus according to any one of claims 2 through 4, wherein said predetermined time point is defined based on a revolution speed of a member that is subjected to a change in revolution speed by speed changing.

8. A control apparatus according to claim 7, wherein said revolution speed is at least one of a revolution speed of an input shaft (1a) of said transmission (1), a revolution speed of an output shaft (1b) of said transmission (1) and a revolution speed of an engine (2), and a vehicle speed.

9. A control apparatus according to claim 7 or claim 8, wherein a defined value of said revolution speed is set according to at least one of an engine load, a vehicle speed and a kind or mode of speed changing.

10. A control apparatus according to any one of claims 2 through 4, wherein said predetermined time point is defined by a torque of an output shaft (1b) of said transmission (1).

11. A control apparatus according to claim 10, wherein a defined value of said torque of said output shaft (1b) is set according to at least one of an engine load, a vehicle speed and kind or mode of transmission.

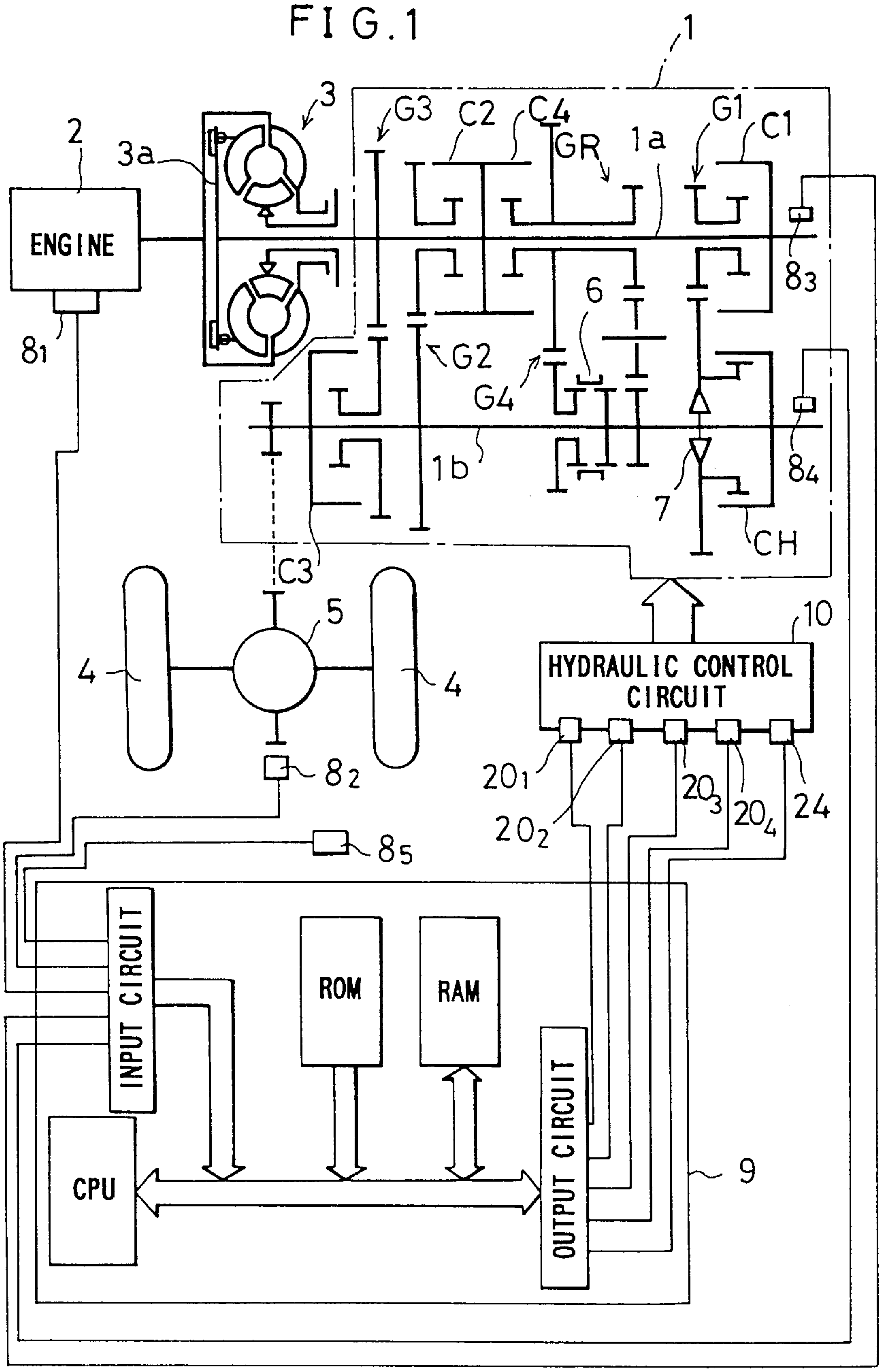
12. A control apparatus according to any one of claims 2 through 4, wherein said predetermined time point is defined based on a hydraulic oil pressure of a third hydraulic engaging element that is engaged during speed changing by disengagement of said second hydraulic engaging element.

13. A control apparatus according to any one of claims 1 through 12, further comprising speed change expecting means for expecting an occurrence of said speed changing, such that said control valve (22) is set to said second position before said speed change judgement or said speed change order.

14. A control apparatus according to claim 13, wherein said speed change expecting means is a timer for detecting a time that is earlier by a predetermined time than the time of said speed change judgement or said speed change order.

15. A control apparatus according to claim 14, wherein a value of said timer is set according to at least one of the engine load, the vehicle speed and the kind or mode of speed changing.

FIG.1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3a
G3
C2
C4
GR
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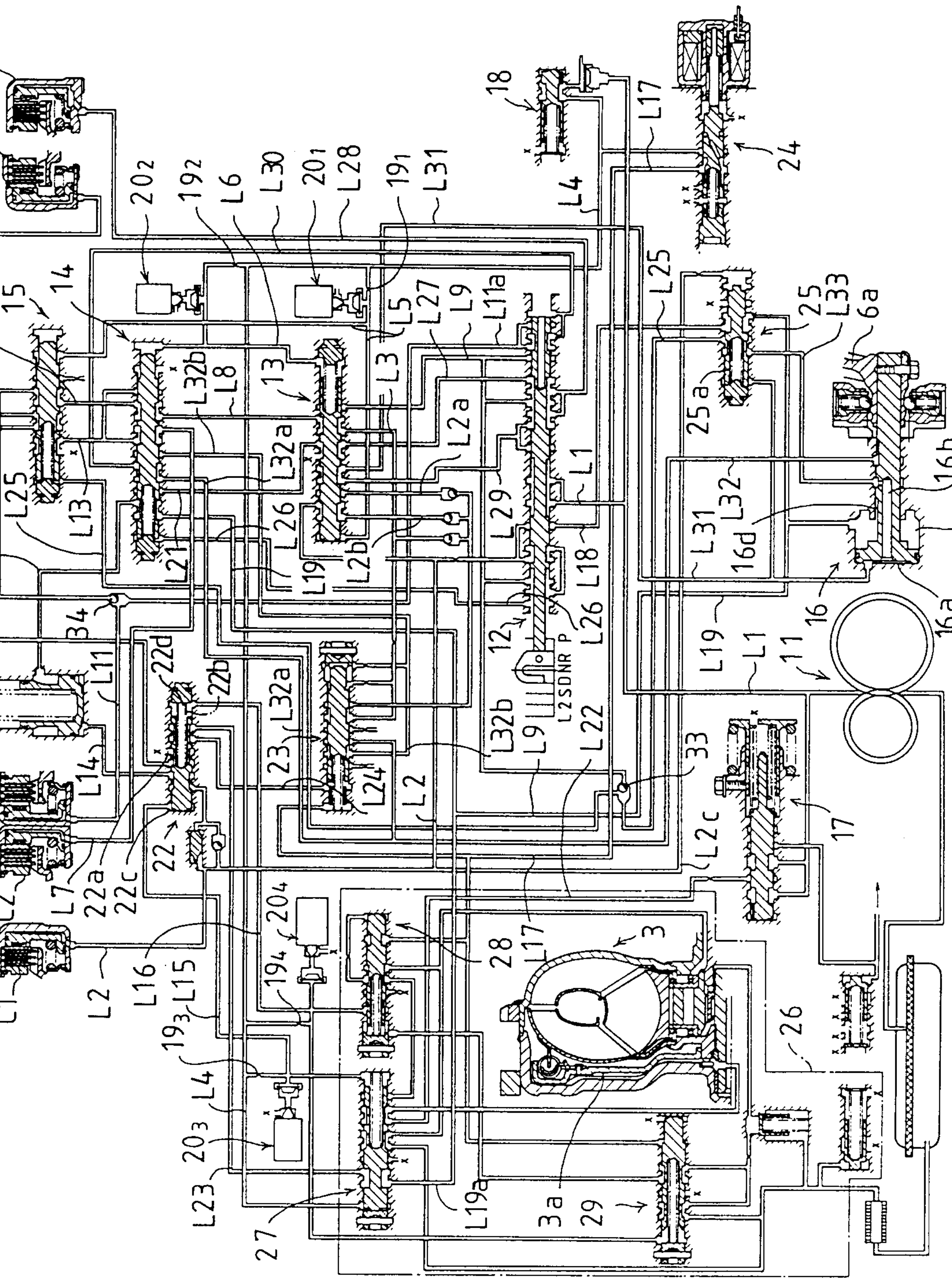
G1
C1
ENGINE
8_3
6
8_1
G2
G4
8_4
1b
7
CH
C3
5
10
4
4
HYDRAULIC CONTROL CIRCUIT
8_2
20_1
20_3
20_4
24
20_2
8_5
INPUT CIRCUIT
ROM
RAM
OUTPUT CIRCUIT
CPU
9

FIG. 2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C3
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LDa
L12
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L6
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20₁
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L27
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L25
25
L33
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25a
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L3
L32b
L8
L32a
L2a
L1
16b
L32
16c
L29
L31
16d
L20
L25
L13
L21
L26
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L2b
L18
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12
L26
16
16a
L19
L1
11
L2 S D N R P
22d
22b
L11
21
L32a
L32b
L9
L22
33
23
L24
L14
C4
C2
L7
22a
22c
22
L2
L2c
17
L17
3
28
20₄
19₄
C1
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L16
L15
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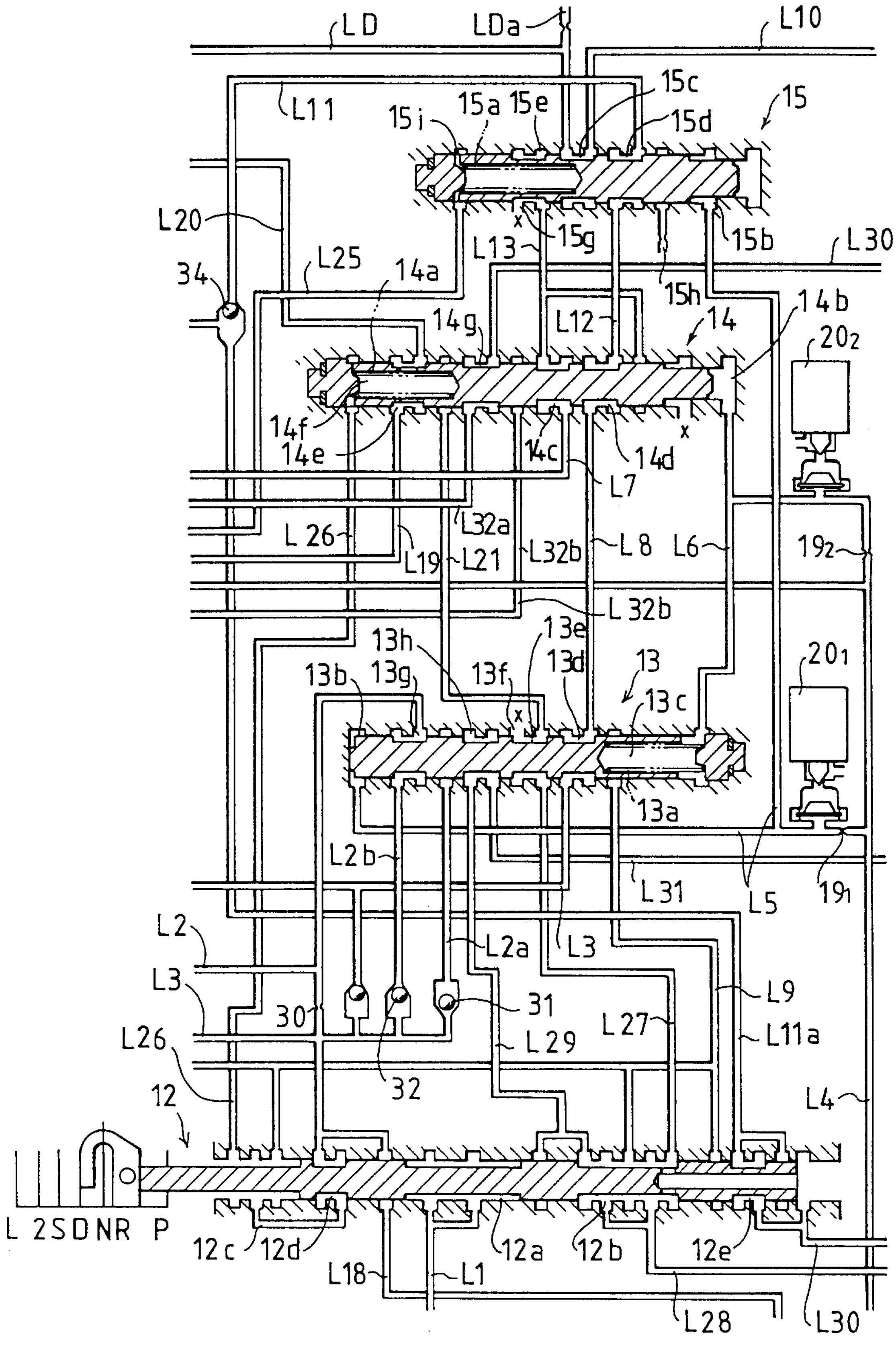
L4
20₃
L23
27
L19a
3a
29

FIG.3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15c
15d
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L20
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14g
L12
14
14b
20₂
14f
14e
14c
14d
L7
L32a
L26
L19
L21
L32b
L8
L6
19₂
L32b
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13e
13b
13g
13f
13d
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13c
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13a
L2b
L31
L5
19₁
L2
L2a
L3
L3
30
31
L9
L27
L11a
L26
L29
32
12
L4
L 2SDNR P
12c
12d
12a
12b
12e
L18
L1
L28
L30

FIG. 4

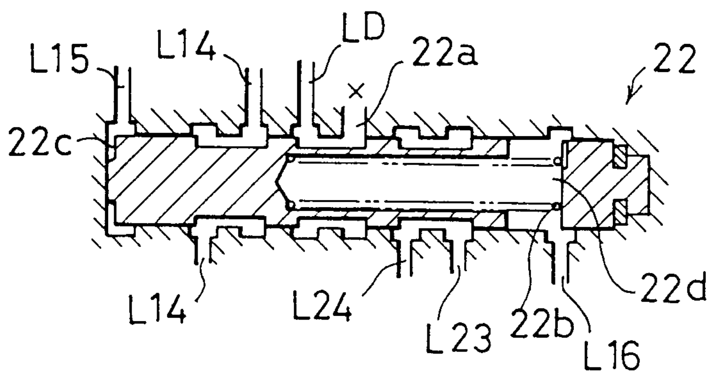


FIG.5A

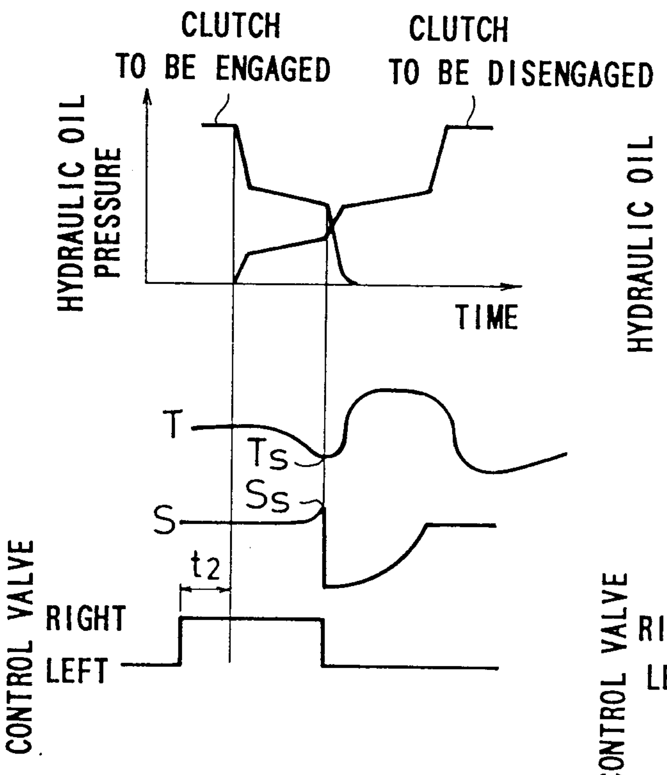
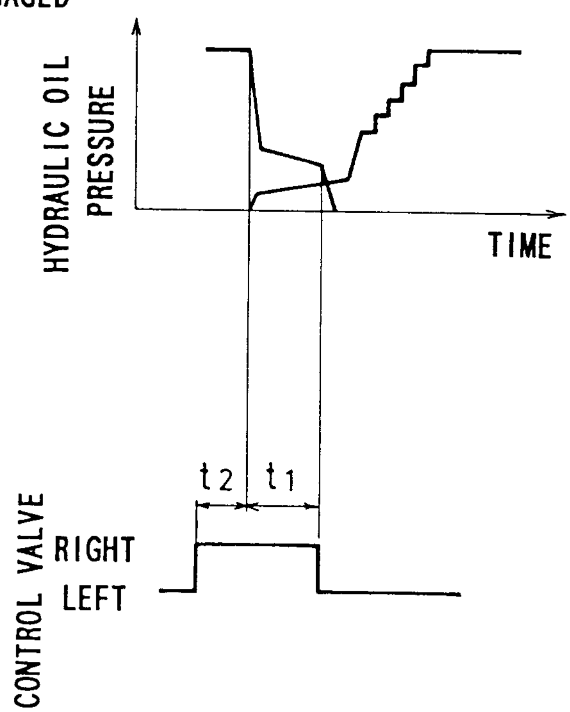


FIG.5B



European Patent Office

EUROPEAN SEARCH REPORT

Application Number

EP 94112435.6

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl. 6)
A	EP - A - 0 133 184 (NISSAN) * Totality; esp. page 1, line 21 - page 3, line 4; claims * --	1	F 16 H 61/06 F 16 H 61/02
A	EP - A - 0 525 725 (HONDA) * Totality; esp. abstract * --	1	
A	EP - A - 0 307 511 (NISSAN) * Totality * --	1	
A	PATENT ABSTRACTS OF JAPAN, unexamined applications, M field, vol. 15, no. 475, December 3, 1991 THE PATENT OFFICE JAPANESE GOVERNMENT page 85 M 1186; & JP-A-03 204 453 (AISIN) --	1	**TECHNICAL FIELDS SEARCHED (Int. Cl.6)**
A	US - A - 5 113 724 (HAYASAKI) * Totality * --	1	B 60 K 41/00 F 16 H 61/00
A	DE - A - 2 931 828 (NISSAN) * Totality * ----	1	

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
VIENNA	21-11-1994	WERDECKER

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
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D : document cited in the application
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